HEAT TRANSFER COEFFICIENTS IN THE BOILING SECTION OF A LONG-TUBE, NATURAL-CIRCULATION EVAPORATOR

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Heat Transfer Coefficients in the Boiling Section of a Long-Tube, Natural-Circulation Evaporator[1]

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THE long-tube, natural-circulation evaporator falls into the classification of the low-level, climbing-film machines. The most common representative is the so-called Kestner evaporator. This particular type was developed in Europe, beginning about 1900 (*17*). For some reason little interest was shown in this country until recently. Because of the redevelopment of this type of machine in America, entirely independent of the old Kestner evaporator, there is now an inclination to speak of it as the LTV (long-tube vertical) evaporator.

The characteristic feature of this type of evaporator is the length of the vertical tubes, up to 22 feet. In operation, the liquid is fed to the bottom of the tubes so slowly that it does not issue from the top as a solid stream but boils in the tube and is carried to the top as a spray or a climbing film.

Previous Work

A number of papers in the foreign literature extol the virtues of the long-tube, natural-circulation type; among its advantages may be mentioned:

1. Ability to handle heat-sensitive liquids without damage (*2, 15, 23, 24, 27, 28*).
2. Relative freedom from scaling trouble (*3, 24, 27, 28*).
3. Freedom from foaming (*1, 15, 24, 27*).

Within the last few years in this country the LTV evaporator has come into some prominence for the handling of materials such as glue, sugar sirup, sulfate waste liquor, etc., of relatively small unit value but with high viscosities. The latter property makes them difficult to handle in the standard types of natural-circulation evaporators, and the pumping cost is a disadvantage of the forced-circulation evaporator. It has been found that all of these materials, in addition to dilute salt brine, may be handled economically in this type of machine (*26*).

Although many papers have been written about the Kestner evaporator, quantitative data on the subject are lacking. The only two papers that present experimental results (*11, 16*) report them in such a form that they cannot be interpreted.

The other types of natural-circulation evaporators have been more thoroughly investigated (*4, 6, 10, 18, 19, 21*). The coefficients reported, either over-all or film, are based on the entire length of the evaporator tube. They are usually spoken of as boiling coefficients but include the effect of a certain but unknown length of tube in which the liquid is not boiling.

This type of coefficient has been found to be a function of the temperature difference (*6, 19, 21*), the liquid level (*4, 6*), the temperature level (*19, 21*), and the concentration of the solution (*21*). In some cases the velocity of circulation was

[1] Complete tables of data obtained in these experiments will be published in the *Transactions of the American Institute of Chemical Engineers.*

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determined (*10*, *19*), but its effect on the coefficient could not be determined since the rate of circulation was a function of other variables.

An exhaustive investigation on an evaporator with a single vertical tube is reported by Kirschbaum, Kranz, and Starck (*18*). In this study the rate of circulation, the liquid temperature gradient along and across the tube, and the tube-wall temperature gradient were determined. A correlation of their data is presented. It is entirely empirical and is in no way general.

Kirschbaum and his coworkers, however, did realize that "the way to the development of the final equation probably leads to a division of the heating surface into two parts, one for heating the liquid and one for evaporation."

Considerable study has been made of the individual coefficients of heat transfer between metal surfaces (internally heated) and boiling liquids. In general, the surface has been a flat plate in either a vertical or a horizontal position (*9*, *12*, *13*, *22*). This work is of interest but as yet shows too great inconsistencies to be of any practical significance.

A method, as suggested by Boarts (*8*), has been developed for determining the length of the boiling section in an externally heated tube.

An empirical correlation of over-all coefficients of heat transfer in the boiling section of a 2-inch o. d., No. 10 Birmingham wire gage, copper tube shows that the coefficient is a function of the amount of vapor present, the weight rate of liquid fed to the tube, and the average over-all temperature difference between the steam and the liquid. This correlation is applicable only within the ranges of variables used for experimentation and to water, or to materials with similar physical properties.

The mechanism of boiling in an externally heated tube is exceedingly complicated. Three types of action, froth, slug, and film, have been found, but no definite connection between the types of action and the conditions of operation could be developed.

Experimental Apparatus

The investigation reported in this paper was carried out in a single-tube evaporator of the type known in commercial practice as the long-tube, natural-circulation evaporator. A flowsheet of the equipment is given in Figure 1:

Distilled water for feed was drawn from storage tanks into the feed weigh tanks, 1, from which the feed pump, 2, forced it into the double-pipe feed heater, 3. An equivalent quantity of water was thus forced into the 2.002 inch o. d. × 1.760 inch i. d. copper tube, 6, which was 20.0 feet long between tube sheets. As the water flowed up the tube, it was heated by the condensation of steam in steam chest 7 until it began to boil. The vapor and liquid were separated by the deflector in vapor head 9. The vapor was led through the vapor line, 16, to condenser, 10, where it was condensed. The unevaporated water left the vapor head through the thick liquor line, 17, going to the thick liquor tanks, 11. The steam condensed in steam chest 7 flowed through the condensate U-bend, 14, into steam drip tanks, 8.

The pressure in vapor head 9 was maintained constant within ±1 mm. of mercury by the manual and solenoid vacuum breaks, 23, 22. The steam pressure was maintained within the same limits by means of the steam control valves, 24, and the steam chest vent line, 15.

Measurements and Control Devices

All temperatures were determined by means of copper-constantan thermocouples. With the exception of the traveling thermocouple, the couples were inserted by sealing the wires, with litharge and glycerol cement, into a 6-inch length of $^3/_8$-inch copper tubing which passed through a standard $^3/_8$-inch compression fitting, drilled out with a No. W drill.

Couples of this type were used to determine the temperatures of the inlet to the evaporator tube, the unevaporated water outlet

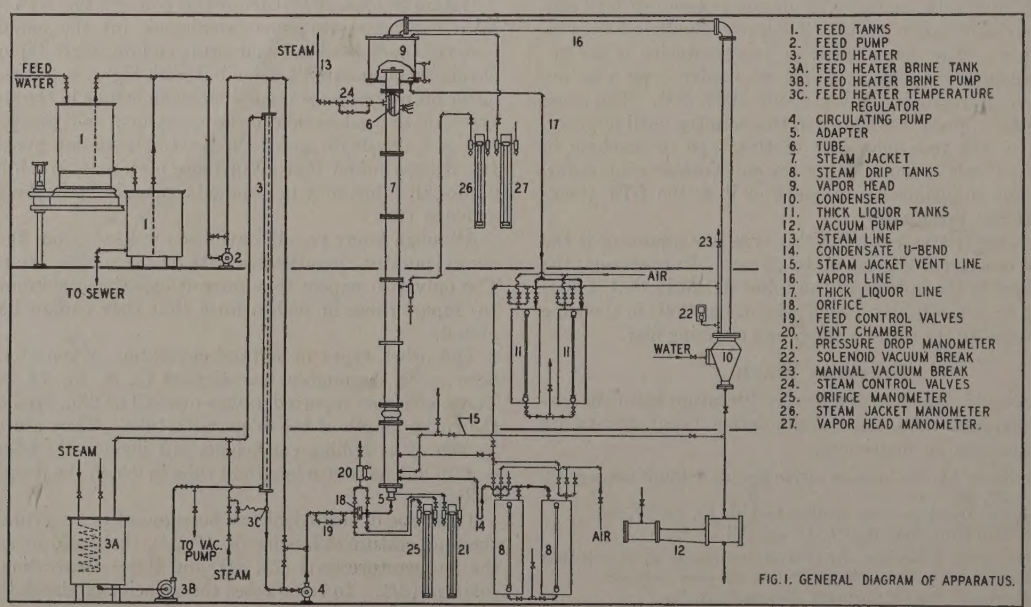


FIG. 1. GENERAL DIAGRAM OF APPARATUS.

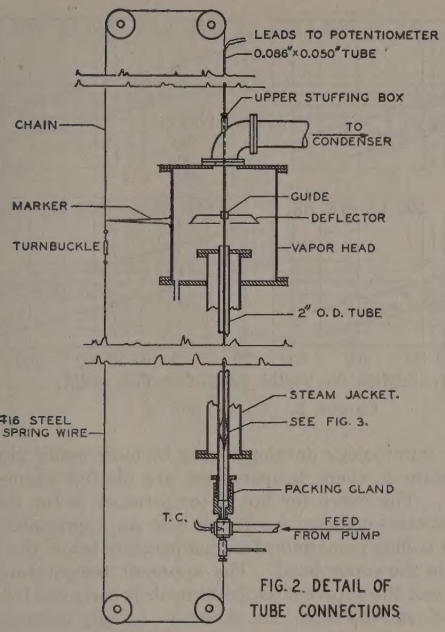


FIG. 2. DETAIL OF TUBE CONNECTIONS

from the vapor head, the vapor in the vapor head, the steam entering the steam chest, the steam in the steam chest (at the top, middle, and bottom), and the condensate leaving the steam chest.

These thermocouples were calibrated in an oil bath against a calibrated platinum resistance thermometer (*5*). The e. m. f. was read with a Leeds & Northrup portable precision potentiometer. The limit of error of the calibration was ±0.04° F.

The traveling couple (Figure 2) for the determination of the liquid temperature along the axis of the tube was of the same general type as that used previously (*8*, *20*). No. 30 Birmingham wire gage, enameled, cotton-covered copper and constantan wires were inserted into an 0.086 inch o. d. × 0.050 inch i. d. tube, 22 feet long. Provision was made for centering the lower end of the tube by vertical fins. The junction extended three-fourths inch beyond the lower end of the brass tube. The leads were withdrawn at the upper end.

The traveling couple was calibrated by comparing the reading in its highest position with the temperature calculated from the vapor head manometer. Conditions in the evaporator were maintained such that fairly vigorous boiling occurred. It was found that the steam temperature could be varied within rather wide limits without appreciable change in the reading of the traveling couple. The limit of error of this calibration was ±0.10° F.

The rate of feed to the evaporator was determined by means of the orifice installed between the control valves and the evaporator, as shown in Figure 1. The total quantity of feed delivered during a run was determined by difference at the feed weigh tanks.

The steam chest and vapor head pressures were determined by means of mercury manometers. All manometer lines were of $^1/_2$-inch standard pipe.

Experimental Procedure

The following ranges of variables were studied:

Variable	Range
Feed rate to the tube, lb./hr.	250, 500, 750, 1000
Feed temp. below vapor head temp., ° F.	10, 25, 30, 50, 60
Boiling point in vapor head, ° F.	150, 175, 200
Steam temp. minus vapor head temp., ° F.	5–50

A run consisted of continuous operation under constant conditions for 20 minutes. All drip tanks and manometers were read at regular intervals, the minimum number of readings being six. Difference readings on the tanks were determined during the run to ensure constancy of heat transfer rate.

Three complete sets of thermocouple readings and three traverses of the tube with the traveling thermocouple were made. The values used for calculation were the average of the three readings in each case.

The total heat input was used as the basis for calculation. Radiation was neglected. Slight superheat was maintained on the inlet steam. The increase in the sensible heat of the feed in being raised from the feed temperature to the maximum liquid temperature was subtracted from the total heat to give the heat transferred through the heating surface in the boiling section, $(Q/\Theta)_B$. This does not include the amount of heat produced by the decrease in the sensible heat content of the unevaporated liquid in cooling from the maximum liquid temperature to the vapor head temperature.

The remaining items in the determination of the over-all heat transfer coefficient in the boiling section are fully discussed below.

In all investigations of this type the condition of the surfaces of the tube is important. An attempt was made in this work to duplicate commercial conditions as closely as possible. Approximately one hundred and fifty runs were made before the data used for this paper were taken. Steam from the University of Michigan powerhouse was used for the heating medium. Hence, the steam side must have closely duplicated the situation present in commercial machines. The condition of the liquid side was controlled by the following procedure: With a given set of feed rate, feed temperature, and boiling point conditions, three runs were made at different values of the apparent temperature difference, spread over the entire range to be covered. Before check runs were made at intermediate values of the temperature difference, the tube was boiled out with a dilute solution of inhibited hydrochloric acid. The results were required to be within ±10 per cent of a smooth curve to be satisfactory.

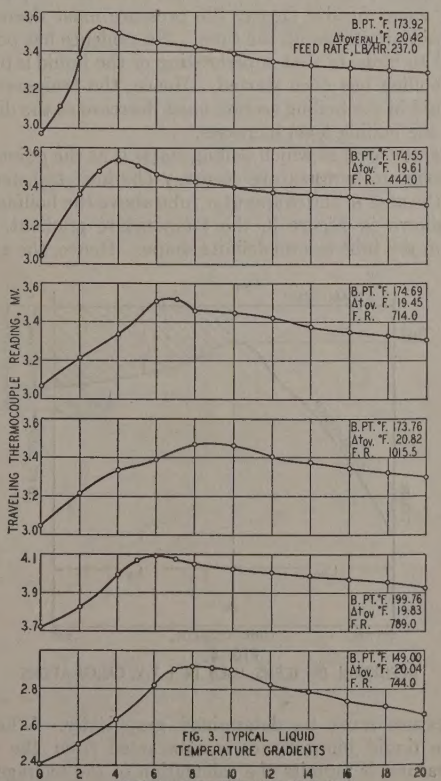


FIG. 3. TYPICAL LIQUID TEMPERATURE GRADIENTS

Determination of Boiling Section

As pointed out above, the liquid temperature distribution along the tube was determined during each run. Several typical curves for temperature *vs.* length are shown in Figure 3. The outstanding characteristic of all of these curves is the maximum which is reached.

The length of the boiling section or the boiling length, L_B, was taken as the length between the point in the tube at which the maximum temperature occurred, and the upper tube sheet. The choice of the maximum liquid temperature as the division point between the section of the tube used for heating and that in which boiling occurs, needs explanation.

The liquid entering the tube, always below the boiling point in the vapor head during this investigation, will be heated as it flows up the tube. At the same time the pressure exerted by the material farther up in the tube (both from static head and friction) will decrease. At some point in the tube, provided boiling occurs, the temperature of the liquid will be such that its vapor pressure will be equal to the pressure existing at that point.

If the liquid comes to equilibrium immediately, boiling should occur when the vapor pressure of the liquid equals the static pressure. There is, however, some doubt that equilibrium is reached at once. Jakob and Fritz (*14*) and Stewart and Hechler (*25*) showed that it is possible to have superheating of the liquid before boiling starts. If this occurs, the liquid will continue up the tube until some temperature is reached, at which point vaporization will occur.

The vapor produced when the liquid boils flows up the tube to the vapor head. Hence, the pressure must decrease as the liquid progresses up the tube. No evidence has been advanced to indicate that superheating of the liquid is possible when boiling has once started. Hence, the temperature of the liquid in the boiling section must decrease as the distance above the boiling level increases.

Thus, the level at which boiling starts is at the point where the maximum temperature occurs. The heat transfer area, A_B, is the area of the evaporator tube above the boiling level.

As shown in Figure 3, the temperature gradient of the liquid in the tube has no definite shape. Hence, the average

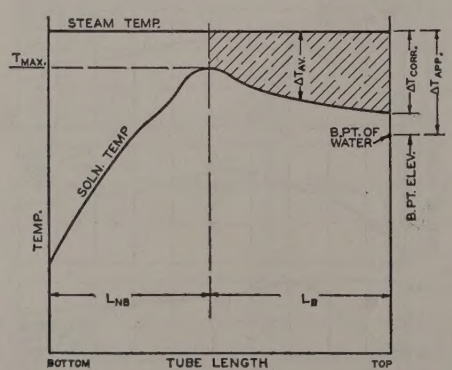


FIG. 4.
DEFINITION OF TERMS USED IN L.T.V. CALCULATIONS

temperature must be determined graphically. When the average liquid temperature is subtracted from the steam temperature, it permits the calculation of the average overall temperature difference in the boiling section, $\Delta t_{av.}$.

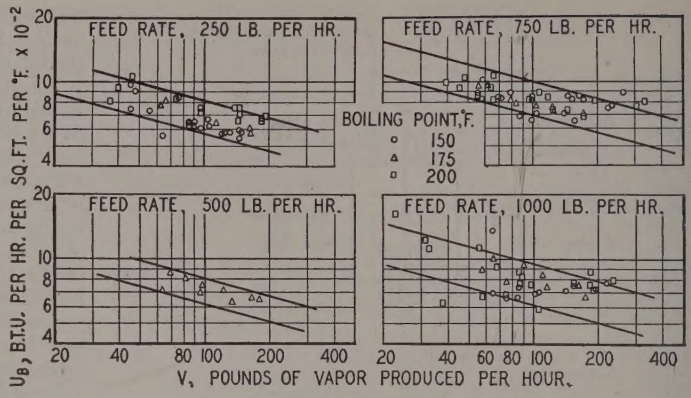

FEED RATE, 250 LB. PER HR.
FEED RATE, 750 LB. PER HR.
FEED RATE, 500 LB. PER HR.
FEED RATE, 1000 LB. PER HR.
BOILING POINT, F.
150
175
200
U_B, B.T.U. PER HR. PER SQ. FT. PER °F. X 10^{-2}
V, POUNDS OF VAPOR PRODUCED PER HOUR.

FIGURE 5. U_B vs. V

The terminology developed may be more easily visualized by Figure 4 where temperatures are plotted against tube length. The curve for liquid temperature is for the most general case—namely, a solution with an appreciable elevation in boiling point and a feed temperature below the boiling point in the vapor head. The apparent temperature drop, $\Delta t_{app.}$, and the apparent temperature drop corrected for boiling point elevation, $\Delta t_{corr.}$, are shown as usually defined. The true mean temperature drop in the boiling section, $\Delta t_{av.}$, is the mean height of the shaded portion. The true boiling coefficient, U_B, reported in this paper is calculated on the surface corresponding to the length, L_B, and on the mean temperature drop, $\Delta t_{av.}$.

Correlation of Results

U_B, the average over-all heat transfer coefficient for the boiling section, is mathematically defined as follows:

$$U_B = \frac{(Q/\Theta)_B}{A_B \Delta t_{av.}}$$

It was believed that the most important variable affecting the boiling heat transfer coefficient must be the velocity of the liquid. It was further believed that the controlling factor in this velocity would be the quantity of vapor present in the boiling section. As the most practicable approach, the total quantity of vapor was used. U_B is plotted against V, the pounds of vapor produced per hour, in Figure 5.

The average slope of the three groups containing the largest number of points was found to be -0.27. This average was determined by averaging the slope of the lines through the groups of points at a constant feed temperature and feed rate, as well as the slopes of the bands.

The remaining variables, W_f (the feed rate to the tube), $\Delta t_{av.}$, L_B, and the temperature level were similarly investigated. The latter two had no appreciable effect.

As shown in Figure 6, it was found that:

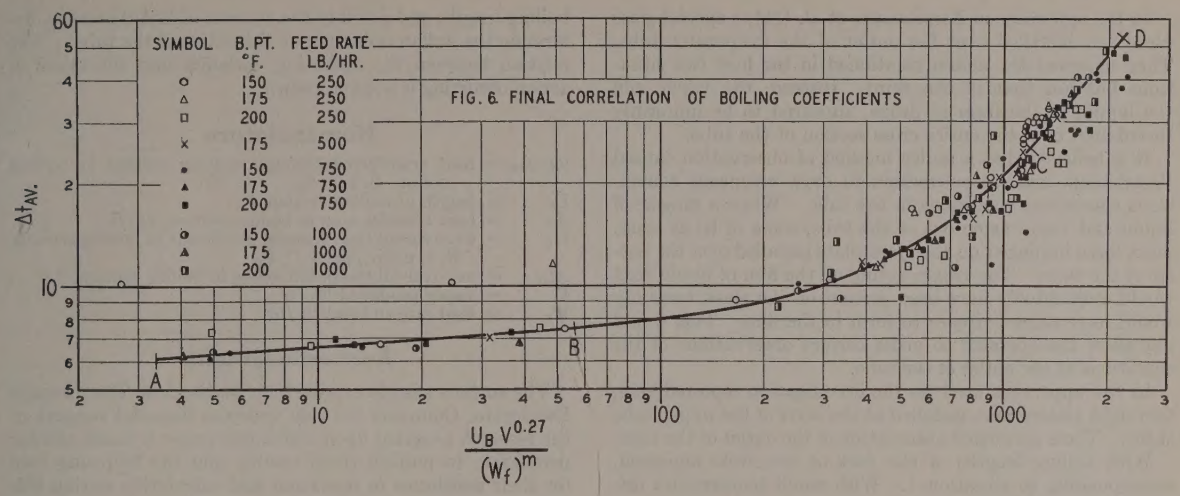
$$\frac{U_B V^{0.27}}{(W_f)^m} = f(\Delta t_{av.})$$

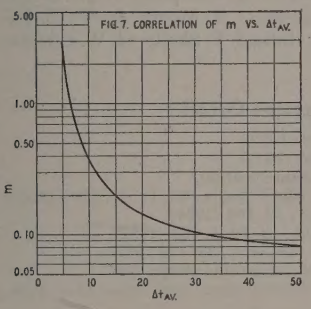
$$m = 0.0557 \log_{10}{}^{-1} \frac{8.28}{\Delta t_{av.}}$$

m is shown graphically as a function of $\Delta t_{av.}$ in Figure 7.

In 81 per cent of the 135 runs the calculated coefficient is within ± 20 per cent of the observed.

It is to be emphasized that this correlation is entirely empirical. The information thus far obtained is not sufficient to permit the proposal of any general, theoretical equation for this type of heat transfer. Because of its empirical nature, this correlation should not be used for design purposes





where the operating conditions are outside the range of variables used during the experiments upon which it is based.

Owing to the nonuniformity of the liquid temperature distribution curves (Figure 3) it is impossible to predict these curves. In order to predict the value of $\Delta t_{av.}$ from the steam temperature and the vapor head temperature, the curves in Figure 8 were developed. In 85 per cent of the runs the calculated value is within ±0.5° F. of the observed value. This is within the limits with which design data are usually known.

Discussion of Results

This investigation is unique in that the heat transfer coefficients reported apply only to the boiling section. Previously reported results have combined the boiling and nonboiling (or heating) sections; the coefficients reported are a weighted average of the coefficients which apply over the two sections.

The method used to determine the boundary between the two sections is that suggested by Boarts (*8*). However, his results of this type were too few to permit correlation. In addition, they are completely out of the range of the present experiments.

Inasmuch as this investigation was concerned primarily with the section in which boiling occurs, attempts were made to determine the phenomena in this section. The literature is deficient in this respect, yielding only two articles in which any attempt was made to explain the mechanism in the boiling section (*7*, *18*).

Barbet (*7*) describes in detail the appearance of the liquid when boiling occurs. He states that with low inlet velocities of the order of 0.3 foot per second, bubbles form which coalesce, pushing slugs of liquid ahead of them. Because of attrition at the walls of the tube, these slugs become thinner and finally break, allowing the vapor to pass through. If vaporization is sufficient, the liquid on the walls of the tube moves upward. Mention was made that annular rings of liquid appeared on the wall of the tube, above the point where the slugs broke.

These experiments were repeated during this investigation, using a glass tube, approximately 3/8 inch o. d., in a glass steam jacket. Barbet's results were verified; i. e., progressing up the tube the following types of action were encountered:

1. A very short length in which the vapor and liquid were more or less uniformly distributed both transversely and longitudinally in the tube.
2. A section of slug action in which the transverse distribution was uniform but the longitudinal was not; i. e., there were alternate slugs of liquid and vapor.
3. A section in which film action occurred, or the longitudinal distribution was uniform, but the transverse was not; i. e., there was a core of vapor with the liquid in a film of varying thickness on the walls of the tube. The annular rings mentioned by Barbet also appeared in this section.

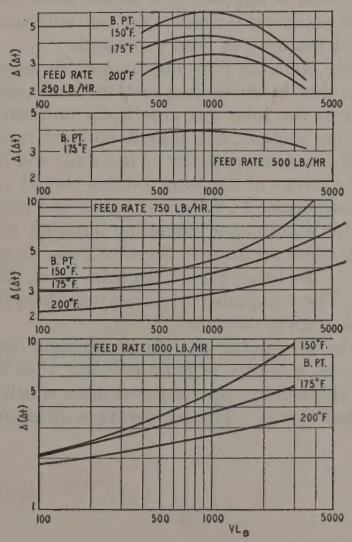


FIGURE 8. CORRELATION OF $\Delta(\Delta t)$ FOR VARIOUS FEED RATES

Correction to over-all temperature difference for the boiling section of 2-inch o. d. tubes. $\Delta(\Delta t)$ = ($\Delta t_{over\text{-}all} - \Delta t_{av.}$) ° F. $\Delta t_{over\text{-}all}$ = (steam temperature − vapor head temperature) ° F. $\Delta t_{av.}$ = (steam temperature − average liquid temperature) °F. L_B = length of boiling section, feet. V = pounds of vapor per hour.

In the apparatus of Kirschbaum et al. (*18*), a special glass plate was installed over the outlet of the evaporator tube. They observed the action mentioned in the first two situations but not that in the third. Instead, the vapor and the liquid, in the form of drops, appeared to be uniformly distributed over the entire cross section of the tube.

It is believed that a faulty method of observation caused Kirschbaum and his co-workers to draw erroneous conclusions concerning conditions in the tube. When a stream of liquid and vapor came out of the tube, some of it, at least, must have impinged on the glass plate installed over the outlet of the tube. The vapor, acting on the film of liquid that would undoubtedly have been present on the glass, certainly would have caused ripples to form in the film. This would not allow the operator to make correct observations of the conditions at the outlet of the tube.

In the apparatus used for the investigation reported here, two sight glasses were installed at the level of the upper tube sheet. These permitted observation of the outlet of the tube.

With boiling lengths of one foot or less, froth appeared, corresponding to situation 1. With small temperature differences the liquid came out in slugs, traveling at high velocity with quiet intervals between the slugs. This corresponds to the second type of action.

As the temperature difference and boiling length increased, two types of spray, coarse and fine, began to appear. The coarse spray seemed to be traveling at a much lower velocity than the fine spray. In addition, the fine spray seemed to be in the center of the tube, whereas the coarse was on the outside. Furthermore, the coarse spray appeared in most cases to be blown outward, but the fine spray traveled up to the deflector in straight lines.

If there were a film on the walls of the tube, it would tend to be blown outward in the form of spray as soon as the retaining walls of the tube were removed. On the other hand, if there were drops of liquid in the vapor stream, they would tend to travel in a straight line for a time, owing to their momentum, even though the line of motion of the vapor changed. Hence, it appears that action according to situation 3 does actually take place with the addition of a phenomenon that has not yet been explained.

In the study made by Kirschbaum and his associates (*18*), it was shown that in the boiling section the temperature distribution across the tube was practically uniform for almost the entire cross section, but that it climbed rapidly at the walls. Since the static pressure at the walls will be the same as that at the center of the tube, boiling should occur at the wall. The formation of a bubble of vapor and its subsequent release into the main vapor stream by bursting, undoubtedly would tend to produce drops. Some of these drops might be thrown over to the other side of the tube—i. e., back into the liquid film. The remainder might stay in the vapor stream to form the fine spray observed.

Hence, it would appear that in progressing upward from the boiling level, a section of froth is found, then slug action, followed by film action, superimposed upon which there may be more or less spray. The temperature difference and the boiling length, and possibly the amount of liquid present, determine the action occurring at the outlet of the tube. The relation between the operating variables and the types of action occurring is not yet known.

Nomenclature

$(Q/\Theta)_B$ = heat transferred through heating surface in boiling section, B. t. u./hr.
L_B = length of boiling section, ft.
A_B = heat transfer area in boiling section, sq. ft.
U_B = av. over-all heat transfer coefficient in boiling section, B. t. u./hr./sq. ft./° F.
$\Delta t_{av.}$ = av. over-all temp. difference in boiling section, ° F.
V = vapor produced, lb./hr.
W_f = feed rate to tube, lb./hr.

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